Representation of Dynamic Performance of Fixed-tilting Pad Mixture Bearing Used In Heavy-duty Gas Turbine *

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Abstract. A novel and universal computational method for obtaining the dynamic stiffness and damping coefficients of the fixed-tilting pad mixture bearing used in heavy-duty gas turbine is given, then the effects of bearing parameters such as eccentricity and perturbation frequencies on those dynamic coefficients are studied.

Introduction

Although fixed-tilting pad mixture bearings are widely used in F type heavy-duty gas turbines[1-3], the theoretical prediction of the dynamic coefficients of fixed-tilting pad mixture bearings is still a difficult problem because of its structural complexity[4]. A fixed-tilting pad mixture bearing as shown in Fig.1 is used in some F type heavy gas turbine. The structure integrates the excellent performance of the upper fixed pad and the two titling ones in order to improve the stability of the bearing. Bearing parameters are expressed as follows.

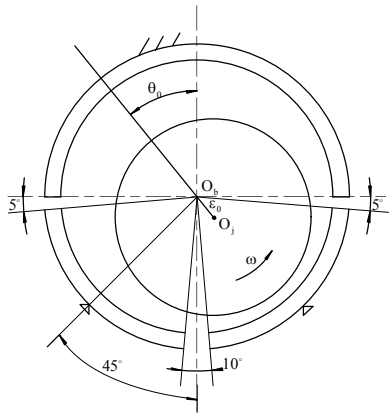


Fig. 1. Structure of fixed-titling pad mixture bearing

Bearing diameter $D = 560mm$, bearing width $L = 448mm$, bearing clearance $C = 0.1mm$. In order to analyze the dynamics of the rotor system，A partial derivative method is used to obtain the stiffness and damping coefficients to characterize dynamic oil film force of the bearing.

Representation of dynamic oil film force of the mixture bearing

Representation of dynamic oil film force of the upper fixed pad. Within the linear range, the dynamic oil film force of the upper fixed pad, as well as the general fixed tile bearings, can be characterized using its eight dynamic stiffness and damping coefficients as

$$\begin{pmatrix} F_x \\ F_y \end{pmatrix}_j = \begin{pmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{pmatrix}_j \begin{pmatrix} x \\ y \end{pmatrix}_j + \begin{pmatrix} d_{xx} & d_{xy} \\ d_{yx} & d_{yy} \end{pmatrix}_j \begin{pmatrix} \dot{x} \\ \dot{y} \end{pmatrix}_j \tag{1}$$

Where k_{ij} and $d_{ij}(i, j = x, y)$ express eight linear the stiffness and damping coefficients of the first pad, $x, y, \dot{x}$ and $\dot{y}$ are perturbation displacements and velocities at the *j*th particle, F_{xj} and F_{yj} express dynamic olm film forces at the *j*th particle.

Representation of dynamic oil film force of the tilting pad. Considering the two lower tilting pads, its dynamic oil film force is not only related to the rotor perturbation, but also closely related to the tile angle disturbance around its pivot swing. Radial perturbation displacement of the titling pad is not considered to obtain dynamic coefficients with the same perturbation prequency of the rotor and bearing, by means of partial derivative method. The dynamic bearing oil film force of the *i*th tilting pad can be expressed as

$$\begin{cases} \bar{F}_{xi} = K_{xxi}\bar{X} + K_{xyi}\bar{Y} + D_{xxi}\dot{\bar{X}} + D_{xyi}\dot{\bar{Y}} + K_{x\delta i}\bar{\delta}_{pi} + D_{x\delta i}\dot{\bar{\delta}}_{pi} \\ \bar{F}_{yi} = K_{yxi}\bar{X} + K_{yyi}\bar{Y} + D_{yxi}\dot{\bar{X}} + D_{yyi}\dot{\bar{Y}} + K_{y\delta i}\bar{\delta}_{pi} + D_{y\delta i}\dot{\bar{\delta}}_{pi} \end{cases} \tag{2}$$

where $\bar{X}$ and $\bar{Y}$ denote the dimensionless perturbation displacement in X and Y directions of the journal center, $\bar{F}_{xi}$ and $\bar{F}_{yi}$ denote the dimensionless dynamic oil film forces at *i*th pad. K_{kmi} （*k,m=x,y*） denotes the dimensionless dyanmic stiffness. D_{kmi} （*k,m=x,y*） expresses the dimensionless dyanmic damping. $K_{k\delta i}$（*k =x,y*） is dimensionless dynamic stiffness coefficients of the *i*th pad subject to the pads perturbation , $D_{k\delta i}$ expresses dimensionless dynamic damping coefficients of the *i*th pad subject to the pads perturbation, $\bar{\delta}_{pi}$ express the dimensionless dynamic swing angle of the *i*th pad. Then the fulcrum moment of the *i*th pad can be expressed as

$$\begin{aligned} \bar{J}_p\ddot{\bar{\delta}}_i &= \bar{b}_i\bar{F}_{xi} - \bar{a}_i\bar{F}_{yi} = (\bar{b}_iK_{xxi} - \bar{a}_iK_{yxi})\bar{X} + (\bar{b}_iK_{xyi} - \bar{a}_iK_{yyi})\bar{Y} + (\bar{b}_iD_{xxi} - \bar{a}_iD_{yxi})\dot{\bar{X}} \\ &+ (\bar{b}_iD_{xyi} - \bar{a}_iD_{yyi})\dot{\bar{Y}} + (\bar{b}_iK_{x\delta i} - \bar{a}_iK_{y\delta i})\bar{\delta}_{pi} + (\bar{b}_iD_{x\delta i} - \bar{a}_iD_{y\delta i})\dot{\bar{\delta}}_{pi} \end{aligned} \tag{3}$$

With the following substitution:

$\begin{cases} \bar{b}_i = \cos(\pi - \beta_i) = -\cos\beta_i \\ \bar{a}_i = \sin(\pi - \beta_i) = \sin\beta_i \end{cases}$, where $\bar{J}_p$ is dimonsionless almoment of inertia of the pads, Eq. (3) can be expressed by dynamic coefficients of the pads as

$$\bar{J}_p\ddot{\bar{\delta}}_i + K_{\delta xi}\bar{X} + K_{\delta yi}\bar{Y} + D_{\delta xi}\dot{\bar{X}} + D_{\delta yi}\dot{\bar{Y}} + K_{\delta\delta i}\bar{\delta}_{pi} + D_{\delta\delta i}\dot{\bar{\delta}}_{pi} = 0 \tag{4}$$

Therefore, the dynamic moment stiffness and damping coefficients K_{kmi} and $D_{kmi}(k,m=x,y)$ of the pads can be obtained by solving Eqs. (3) and（4）.

Let $\begin{cases} \overline{X} = \overline{X}_0 e^{i\Omega T} \\ \overline{Y} = \overline{Y}_0 e^{i\Omega T} \\ \overline{\delta}_{pi} = \overline{\delta}_{0i} e^{i\Omega T} \end{cases}$, and the Eqs.（4）is changed as

$$-\overline{J}_p \Omega^2 \overline{\delta}_{0i} = -\left[(K_{\delta xi} + i\Omega D_{\delta xi})\overline{X}_0 + (K_{\delta yi} + i\Omega D_{\delta yi})\overline{Y}_0 + (K_{\delta\delta i} + i\Omega D_{\delta\delta i})\overline{\delta}_{0i} \right]$$

Then

$$\overline{\delta}_{0i} = \frac{-\left[(K_{\delta xi} + i\Omega D_{\delta xi})\overline{X}_0 + (K_{\delta yi} + i\Omega D_{\delta yi})\overline{Y}_0 \right]}{(K_{\delta\delta i} - \overline{J}_p \Omega^2) + i\Omega D_{\delta\delta i}} \tag{5}$$

Where $\Omega = \upsilon / \omega$ represents the dimonsionless perturbation frequency, υ represents the perturbation angular velocity, and ω represents rotating angular velocity of the rotor.

And the dimonsionless dynamic oil film force of the fixed-titling pad mixture bearing can be expressed in the rectangular coordinate system as

$$\begin{cases} \overline{F}_x = \sum_{i=1}^{3} F_{xi} = K_{xx}^{\#}\overline{X} + K_{xy}^{\#}\overline{Y} + D_{xx}^{\#}\dot{\overline{X}} + D_{xy}^{\#}\dot{\overline{Y}} + \sum_{i=2}^{3}(K_{x\delta i}\overline{\delta}_{pi} + D_{x\delta i}\dot{\overline{\delta}}_{pi}) \\ \overline{F}_y = \sum_{i=1}^{3} F_{yi} = K_{yx}^{\#}\overline{X} + K_{yy}^{\#}\overline{Y} + D_{yx}^{\#}\dot{\overline{X}} + D_{yy}^{\#}\dot{\overline{Y}} + \sum_{i=2}^{3}(K_{y\delta i}\overline{\delta}_{pi} + D_{y\delta i}\dot{\overline{\delta}}_{pi}) \end{cases} \tag{6}$$

where $\overline{F}_x$ and $\overline{F}_y$ denote the dimonsionless dynamic oil film force of the bearing in x and y directions with the same pertubation of the rotor and the pads, $K_{km}^{\#} = \sum_{i=1}^{3} K_{kmi}$ and $D_{km}^{\#} = \sum_{i=1}^{3} D_{kmi}$ are the integral dynamic stiffness and damping coefficients of the fixed-titling pad mixture bearing.

Let $\overline{F}_x = F_{x0} e^{i\Omega T}, \overline{F}_y = F_{y0} e^{i\Omega T}$, then the following representation can be got:

$$\begin{cases} F_{x0} = (K_{xx}^{\#} + i\Omega D_{xx}^{\#})\overline{X}_0 + (K_{xy}^{\#} + i\Omega D_{xy}^{\#})\overline{Y}_0 + \sum_{i=1}^{n}(K_{x\delta i} + i\Omega D_{x\delta i})\overline{\delta}_{0i} \\ F_{y0} = (K_{yx}^{\#} + i\Omega D_{yx}^{\#})\overline{X}_0 + (K_{yy}^{\#} + i\Omega D_{yy}^{\#})\overline{Y}_0 + \sum_{i=1}^{n}(K_{y\delta i} + i\Omega D_{y\delta i})\overline{\delta}_{0i} \end{cases} \tag{7}$$

where F_{x0} and F_{y0} denote the ample of dynamic oil film force of the bearing in x and y directions.

Within the linear range, the dynamic oil film force of the bearing is also expressed by eight dynamic stiffness and damping coefficients. If the amount dynamic stiffness and damping

coefficients is separately expressed by K_{km}^* and $D_{km}^*(k,m=x,y)$, the dynamic oil film force of the whole mixture bearing can be expressed as

$$\begin{cases} \bar{F}_x = K_{xx}^*\bar{X} + K_{xy}^*\bar{Y} + D_{xx}^*\dot{\bar{X}} + D_{xy}^*\dot{\bar{Y}} \\ \bar{F}_y = K_{yx}^*\bar{X} + K_{yy}^*\bar{Y} + D_{yx}^*\dot{\bar{X}} + D_{yy}^*\dot{\bar{Y}} \end{cases} \tag{8}$$

so

$$\begin{cases} F_{x0} = (K_{xx}^* + i\Omega D_{xx}^*)\bar{X}_0 + (K_{xy}^* + i\Omega D_{xy}^*)\bar{Y}_0 \\ F_{y0} = (K_{yx}^* + i\Omega D_{yx}^*)\bar{X}_0 + (K_{yy}^* + i\Omega D_{yy}^*)\bar{Y}_0 \end{cases} \tag{9}$$

And then the amount dynamic stiffness and damping coefficients obtained by iteratively solving Eqs. (5), (7) and (9) as follows:

$$\begin{cases} K_{km}^* = K_{km}^{\#} - \sum_{i=1}^{3}\left[p\cdot(K_{k\delta i}K_{\delta mi} - \Omega^2 D_{k\delta i}D_{\delta mi}) + \Omega^2 q\cdot(K_{k\delta i}D_{\delta mi} + K_{\delta mi}D_{k\delta i})\right] \\ D_{km}^* = D_{km}^{\#} - \sum_{i=1}^{3}\left[p\cdot(K_{k\delta i}D_{\delta mi} + K_{\delta mi}D_{k\delta i}) + \Omega^2 q\cdot(D_{k\delta i}D_{\delta mi} - \frac{1}{\Omega^2}K_{k\delta i}K_{\delta mi})\right] \end{cases} \tag{10}$$

where

$$\begin{cases} p = \dfrac{K_{\delta\delta i} - \bar{J}_p\Omega^2}{(K_{\delta\delta i} - \bar{J}_p\Omega^2)^2 + (\Omega D_{\delta\delta i})^2} \\ q = \dfrac{D_{\delta\delta i}}{(K_{\delta\delta i} - \bar{J}_p\Omega^2)^2 + (\Omega D_{\delta\delta i})^2} \end{cases} \tag{11}$$

So the dimension equivalent dynamic stiffness and damping coefficients of the fixed-titling pad mixture bearing can be expressed as

$$\begin{cases} k_{km}^* = \dfrac{2\mu\omega L}{\psi^3}K_{km}^* \\ d_{km}^* = \dfrac{2\mu L}{\psi^3}D_{km}^* \end{cases} \quad (k,\ m = x,y) \tag{12}$$

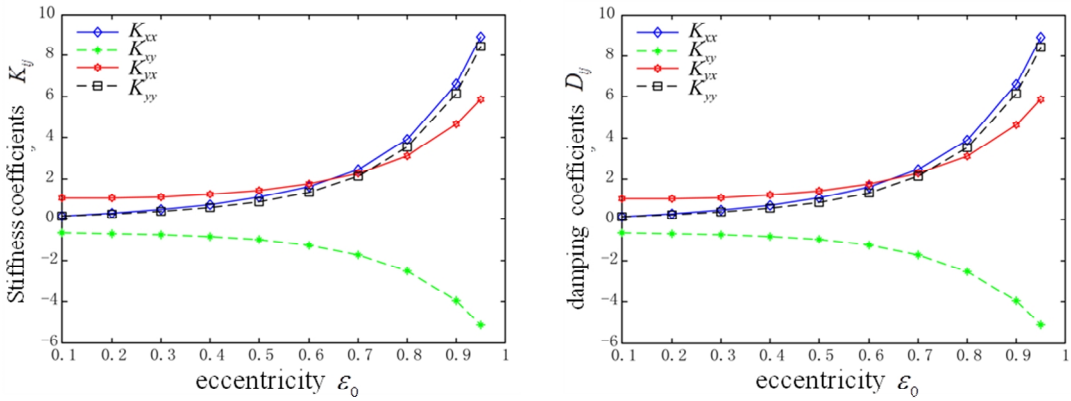
where L denotes bearing width/m, μ denotes dynamic viscosity of ole/Pa·s, ω denotes rotating angular velocity of rotor/rad·s^{-1}, ψ denotes clearance ratio of bearing.

Then the dimension equivalent dynamic stiffness and damping coefficients of the fixed-titling pad mixture bearing can be expressed as

$$\begin{cases} F_x = k_{xx}^* X + k_{xy}^* Y + d_{xx}^* \dot{X} + d_{xy}^* \dot{Y} \\ F_y = k_{yx}^* X + k_{yy}^* Y + d_{yx}^* \dot{X} + d_{yy}^* \dot{Y} \end{cases} \tag{13}$$

Dynamic Performance of the fixed-titling mixture bearing

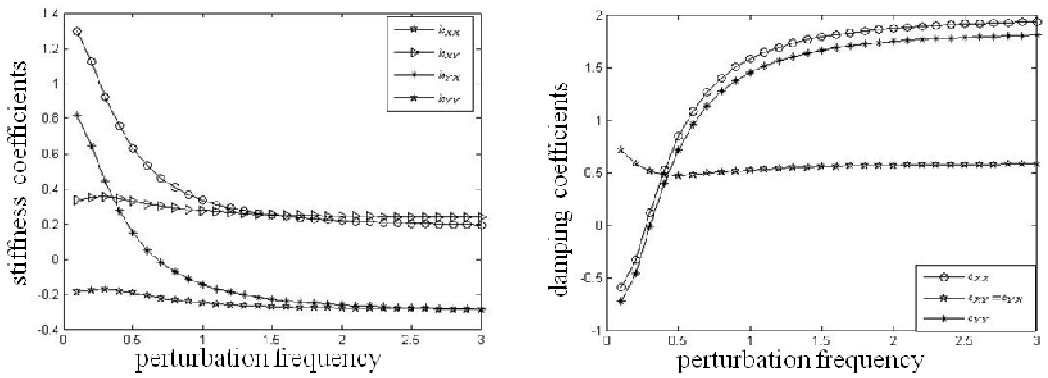
The amount dynamic stiffness and damping coefficients of the mixture bearing is calculated using the proposed method to analysize the influence of the eccentricity and the perturbation frequency upon the results. Fig.2 illustrates the variations of the amount dynamic stiffness and damping coefficients of the mixture bearing with eccentricity for static load at $m = 0.0$. It can be found that the amount dynamic stiffness and damping coefficients increase with ε_0 increasing and the cross damping coefficients have little changes.



(a) dynamic stiffness (b) damping coefficients

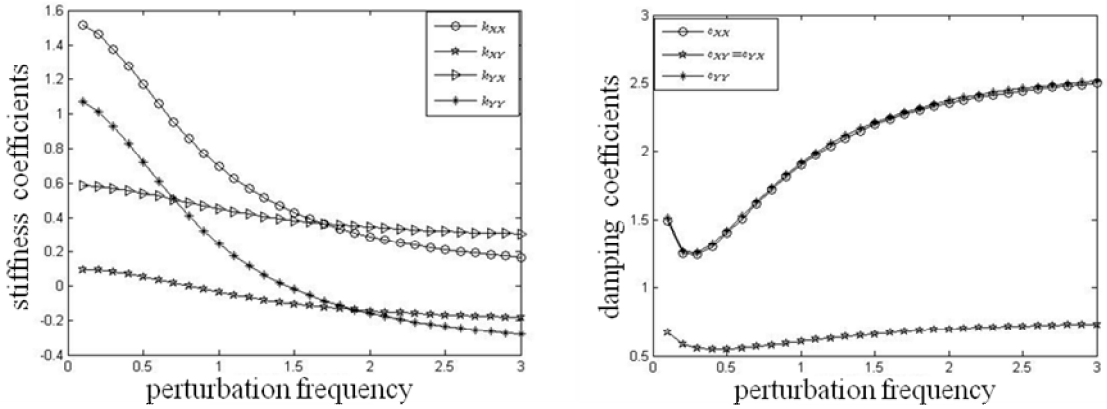
Fig.2 Changes of dynamic stiffness and damping coefficients with the eccentricity at m=0

Fig.3 and Fig.4 depict the influention of the perturbation frequency of the rotor and pads upon the dynamic coefficients. it is observed that the dynamic stiffness and damping coefficients are closely associated with the perturbation frequency. The amount dynamic stiffness coefficients decrease while the amount dynamic damping coefficients increase with perturbation frequency increasing at lower frequencies. The values of dynamic stiffness and damping coefficients become steady with higher perturbation frequency increacing.



(a) dynamic stiffness coefficients (b) dynamic damping coefficients

Fig.3 Changes of dynamic stiffness and damping coefficients with the journal perturbation frequency at $\varepsilon = 0.1$



(a) dynamic stiffness coefficients (b) dynamic damping coefficients

Fig.4 Changes of dynamic stiffness and damping coefficients with the journal perturbation frequency at $\varepsilon = 0.3$

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